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CAM RING BEARING FOR FUEL DELIVERY SYSTEM

Background of the Invention

The present invention relates to a bearing arrangement, and more particularly to a bearing arrangement used to support a cam ring within a support member or yoke in a hydrostatic and hydrodynamic configuration for use in fuel pumps, metering, and control for jet engines.

PCT/US02/09298, filed March 27, 2002, the details of which are incorporated herein by reference, relates to a fuel delivery system having increased efficiency and reliability over known fuel pump arrangements. Particularly, a pump of a fuel delivery system includes a housing having a chamber with an inlet and outlet in fluid communication with the pump chamber. A rotor is received in the pump chamber, and a cam member surrounds the rotor and is freely rotatable relative to the housing and the rotor. A journal bearing is formed between the cam ring and a support sleeve or yoke that is precluded from rotation within the housing.

The bearing arrangement must be responsive to hydrostatic and hydrodynamic forces imposed thereon by the internal components of the pumping mechanism. Known bearing arrangements require improvement to properly support the cam ring in a combined hydrostatic and hydrodynamic arrangement. Accordingly, a need exists for a new bearing assembly.

Summary of the Invention

An improved bearing assembly is provided for a fuel delivery system that includes a housing receiving a rotor within a rotatable cam ring, where the cam ring is freely rotatable relative to the housing and the rotor. The bearing assembly includes an annular surface having a central opening dimensioned to receive the associated cam ring. The annular surface includes a first, high pressure pad and a second low pressure pad spaced by first and second lands.

The circumferential extension of the first pad is at least as great as an inner diameter of the cam ring.

Circumferential ends of the second pad are preferably wider than the circumferential ends of the first pad.

A differential pressure is established across the pump chamber and the cam ring is capable of movement between the high and low pressure pads in response to

pressure variations. Clearance between the land and the cam ring selectively alters the flow of fluid through the bearing to maintain a pressure. This creates a relatively stiff bearing mount without deflection concerns.

A primary advantage of the invention resides in an improved bearing interface between a rotating cam ring and stationary (non-rotatable), but moveable yoke.

Another advantage of the invention resides in the structure being capable of providing hydrostatic bearing capabilities, as well as hydrodynamic bearing capabilities.

Still other benefits and advantages of the invention will become apparent to those skilled in the art upon a reading and understanding of the following detailed description.

Brief Description of the Drawings

Figure 1 is an exploded perspective view of a preferred embodiment of the fluid pump.

Figure 2 is a cross-sectional view through the assembled pump of Figure 1.

Figure 3 is a longitudinal cross-sectional view through the assembled pump.

Figure 4 is a cross-sectional view similar to Figure 2 illustrating a variable displacement pump with the support ring located in a second position.

Figure 5 is an enlarged cross-sectional view of the pump.

Figure 6 is an exploded perspective view of the bearing assembly.

Detailed Description of the Invention

As shown in the Figures, a pump assembly **10** includes a housing **12** having a pump chamber **14** defined therein. Rotatably received in the chamber is a rotor **20** secured to a shaft **22** for rotating the rotor within the chamber. Peripherally or circumferentially spaced about the rotor are a series of radially extending grooves **24** that operatively receive blades or vanes **26** having outer radial tips that extend from the periphery of the rotor. The vanes may vary in number, for example, nine (9) vanes are shown in the embodiment of Figure 2, although a different number of vanes can be used without departing from the scope and intent of the present invention. As is perhaps best

illustrated in Figure 2, the rotational axis of the shaft **22** and rotor **20** is referenced by numeral **30.** Selected vanes (right-hand vanes shown in Figure 2) do not extend outwardly from the periphery of the rotor to as great an extent as the remaining vanes (left-hand vanes in Figure 2) as the rotor rotates within the housing chamber. Pumping chambers are defined between each of the vanes as the vanes rotate in the pump chamber with the rotor and provide positive displacement of the fluid.

With continued reference to Figure 2, a spacer ring **40** is rigidly secured in the housing and received around the rotor at a location spaced adjacent the inner wall of the housing chamber. The spacer ring has a flat or planar cam rolling surface **42** and receives an anti-rotation pin **44**. The pin pivotally receives a cam sleeve **50** that is non-rotatably received around the rotor. First and second lobes or actuating surfaces **52, 54** are provided on the sleeve, typically at a location opposite the anti-rotation pin. The lobes cooperate with first and second actuator assemblies **56, 58** to define means for altering a position of the cam sleeve **50**. The altering means selectively alter the stroke or displacement of the pump in a manner well known in the art. For example, each actuator assembly includes a piston **60**, biasing means such as spring **62**, and a closure member **64** so that in response to pressure applied to a rear face of the pistons, actuating lobes of the cam sleeve are selectively moved. This selective actuation results in rolling movement of the cam sleeve along a generally planar or flat surface **66** located along an inner surface of the spacer ring adjacent on the pin **44**. It is desirable that the cam sleeve undergo a linear translation of the centerpoint, rather than arcuate movement, to limit pressure pulsations that may otherwise arise in seal zones of the assembly. In this manner, the center of the cam sleeve is selectively offset from the rotational axis **30** of the shaft and rotor when one of the actuator assemblies is actuated and moves the cam sleeve (Figure 2). Other details of the cam sleeve, actuating surface, and actuating assemblies are generally well known to those skilled in the art so that further discussion herein is deemed unnecessary.

Received within the cam sleeve is a rotating cam member or ring **70** having a smooth, inner peripheral wall **72** that is contacted by the outer tips of the individual vanes **26** extending from the rotor. An outer, smooth peripheral wall **74** of the cam ring is configured for free rotation within the cam sleeve **50**. More particularly, a journal bearing **80** supports the rotating cam ring **70** within the sleeve. The journal

bearing is filled with the pump fluid, here jet fuel, and defines a hydrostatic or hydrodynamic, or a hybrid hydrostatic/hydrodynamic bearing. The frictional forces developed between the outer tips of the vanes and the rotating cam ring **70** result in a cam ring that rotates at approximately the same speed as the rotor, although the cam ring is free to rotate relative to the rotor since there is no structural component interlocking the cam ring for rotation with the rotor. It will be appreciated that the ring rotates slightly less than the speed of the rotor, or even slightly greater than the speed of the rotor, but due to the support/operation in the fluid film bearing, the cam ring possesses a much lower magnitude viscous drag. The low viscous drag of the cam ring substitutes for the high mechanical losses exhibited by known vane pumps that result from the vane frictional losses contacting the surrounding stationary ring. The drag forces resulting from contact of the vanes with the cam ring are converted directly into mechanical losses that reduce the pumps overall efficiency. The cam ring is supported solely by the journal bearing **80** within the cam sleeve. The journal bearing is a continuous passage. That is, there is no interconnecting structural component such as roller bearings, pins, or the like that would adversely impact on the benefits obtained by the low viscous drag of the cam ring. For example, flooded ball bearings would not exhibit the improved efficiencies offered by the journal bearing, particularly a journal bearing that advantageously uses the pump fluid as the fluid bearing.

In prior applications these mechanical drag losses can far exceed the mechanical power to pump the fluid in many operating regimes of the jet engine fuel pump. As a result, there was a required use of materials having higher durability and wear resistance because of the high velocity and load factors in these vane pumps. The material weight and manufacturing costs were substantially greater, and the materials also suffer from high brittleness. The turning speed of those pumps was also limited due to the high vane sliding velocities relative to the cam ring. Even when using special materials such as tungsten carbide, high speed pump operation, e.g., over 12,000 RPM, was extremely difficult.

These mechanical losses resulting from friction between the vane and cam ring are replaced in the present invention with much lower magnitude viscous drag losses. This results from the ability of the cam ring to rotate with the rotor vanes. A relatively low sliding velocity between the cam ring and vanes results, and allows the manufacturer

to use less expensive, less brittle materials in the pump. This provides for increased reliability and permits the pump to be operated at much higher speeds without the concern for exceeding tip velocity limits. In turn, higher operating speeds result in smaller displacements required for achieving a given flow. In other words, a smaller, more compact pump can provide similar flow results as a prior larger pump. The pump will also have an extended range of application for various vane pump mechanisms.

Figure 3 more particularly illustrates inlet and outlet porting about the rotor for providing an inlet and outlet to the pump chamber. First and second plates **90, 92** have openings **94, 96**, respectively. Energy is imparted to the fluid by the rotating vanes. Jet fuel, for example, is pumped to a desired downstream use at an elevated pressure.

As shown in Figure 4, neither of the actuating assemblies is pressurized so that the cam sleeve is not pivoted to vary the stroke of the vane pump. That is, this no flow position of Figure 4 can be compared to Figure 2 where the cam sleeve **50** is pivoted about the pin **44** so that a close clearance is defined between the cam sleeve and the spacer ring **40** along the left-hand quadrants of the pump as illustrated in the Figure. This provides for variable displacement capabilities in a manner achieved by altering the position of the cam sleeve.

In the preferred arrangement, the vanes are still manufactured from a durable, hard material such as tungsten carbide. The cam ring and side plates, though, are alternately formed of a low cost, durable material such as steel to reduce the weight and manufacturing costs, and allow greater reliability. Of course, it will be realized that if desired, all of the components can still be formed of more expensive durable materials such as tungsten carbide and still achieve substantial efficiency benefits over prior arrangements. By using the jet fuel as the fluid that forms the journal bearing, the benefits of tungsten carbide for selected components and steel for other components of the pump assembly are used to advantage. This is to be contrasted with using oil or similar hydraulic fluids as the journal bearing fluid where it would be necessary for all of the jet fuel components to be formed from steel, thus eliminating the opportunity to obtain the benefits offered by using tungsten carbide.

As illustrated in greater particularity in Figures 5 and 6, the journal bearing assembly defined by the interface between the cam sleeve or yoke **50** and the cam ring **70**

is shown in greater detail. Particularly, the inner surface **100** of the support sleeve or yoke is a non-constant diameter to define discrete portions of the bearing arrangement. Specifically, a first or large diameter portion **102** defines a first, high pressure pad and a diametrically opposite, second or low pressure pad **104**. For ease of description, and as will be appreciated from Figure 5, the high pressure pad portion **102** extends from approximately 4 o'clock to 8 o'clock while the low pressure pad extends from approximately 10 o'clock to 2 o'clock. Separating the high pressure pad from the low pressure pad are first and second seal lands **106, 108**. The first seal land **106**, therefore extends from approximately 2 o'clock to 4 o'clock, while the second seal land **108** extends from approximately 8 o'clock to 10 o'clock.

The bearing arrangement defines a combination hydrostatic and hydrodynamic configuration. The hydrostatic portion of the bearing is the two pad arrangement defined by the high pressure and low pressure pads **102, 104**, respectively. The high pressure pad is a groove cut through the full width or extent of the yoke, i.e., from a front face **50a** to a rear face **50b,** as will be more clearly appreciated from a review of Figure 6. Likewise, the low pressure pad is also a groove through the full width of the yoke. The high pressure pad is capable of supporting the forces generated by the internal components of the pumping mechanism. Between the two pads, in the yoke, are the seal lands **106, 108** that create a hydrodynamic effect that enables smooth start-up and centers the cam ring within the bearing during operation.

The high pressure pad geometry is determined so that the force generated by the fluid pressure is slightly greater than the forces generated by the internal pumping elements. The circumferential extent of the pad **102**, i.e., from 4 o'clock to 8 o'clock, is determined by the radial thickness of the cam ring. It is preferred that the edges **102a, 102b** of the high pressure pad are located outside the inside diameter **72** of the cam ring (see Figure 5). The seals and the sides of the high pressure groove, that is along the faces **50a, 50b** of the yoke, are created by the port plates **90, 92** (Figure 3) which clamp across the pumping element. High pressure fluid (jet fuel) is fed into the pad through openings **120** shown in Figure 6 and the flow to the interface between the yoke and the cam ring is restricted through orifices **122** (only one of which is seen in the view of Figure 6). As will be appreciated, the high pressure orifices **122** communicate with respective openings or holes **120** in this region of the bearing assembly.

The geometry of the low pressure pad **104** is determined by setting circumferential edges **104a, 104b** slightly wider than the circumferential edges of the high pressure pad, i.e., slightly wider than **102a, 102b**, respectively. Venting from the high pressure pad to the low pressure pad must be provided in this pad such that high pressure does not build. This is provided through openings **124**, one of which is illustrated in Figure 6. As will be apparent, openings **124** have a substantially larger diameter than openings **122**. Therefore, a differential pressure is established across the yoke to react the forces within the pumping element.

The high and low pressure pads **102, 104** are cut completely through the bearing, i.e., they extend completely from face **50a** to **50b**, to allow the cam ring to move in the vertical direction as depicted in Figure 5. The movement in the vertical direction allows for radial deflection of the yoke in the horizontal direction, thus increasing the clearance between the lands and the cam ring. When the clearance increases, the flow through the bearing must increase to maintain the pressure in the high pressure pad, or the clearance must be reduced. The orifices **122** on the high pressure pad side restrict the flow and thus the cam ring moves vertically forward decreasing the clearance to re-establish an equilibrium force condition. This creates a relatively stiff bearing without the concerns of deflection.

The entire bearing, yoke **50** and cam ring **70** is free to roll within the pumping mechanism as described above. As shown in Figure 5, the bearing rolls leftwardly or rightwardly along the generally planar surface **42** provided in the spacer ring **40**. This rolling on the surface **42** acts to provide a linear translation of the cam ring. Linear cam ring translation is critical to minimizing fluid pump pressure pulsation during operation. Sliding and rotation of the yoke are prevented by the anti-rotation disks **44** inserted on each side of the yoke. As will be apparent from Figure 6, these anti-rotation disks **44** are dimensioned for receipt in arcuate recesses or cutouts **130**, only one of which is illustrated in Figure 6, although it will be appreciated that a similar cutout recess is provided on the rear surface **50b** of the yoke. Thus, these anti-rotation disks **44** do not pass completely through the yoke, or corresponding recesses provided in the spacer ring, and thereby allow the forces in yoke to be transmitted to the housing structure through the spacer ring.

It will also be appreciated that in the preferred embodiment of the yoke **50**, an undercut **140** is provided on the first and second surfaces **50a, 50b**. The undercut **140** is provided at the outer radial perimeter of these faces. Moreover, the undercut extends circumferentially around substantially the entire yoke, i.e., from approximately 6:30 in a clockwise direction to approximately 5:30. The undercut facilitates control of pressure on the face of the yoke and accurately predicts or controls the pressure of the overall pump arrangement.

The invention has been described with reference to the preferred embodiments. Obviously, modifications and alterations will occur to others upon reading and understanding the preceding detailed description. It is intended that the invention be construed as including all such modifications and alterations in so far as they come within the scope of the appended claims or the equivalents thereof.